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ORIGINAL RESEARCH PAPER

Numerical Study of Single Phase/Two-Phase Models for Nanofluid Forced Convection and Pressure Drop in a Turbulence Pipe Flow

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ABSTRACT: In this paper, the problem of turbulent forced convection flow of water- alumina nanofluid in a uniformly heated pipe has been thoroughly investigated. In numerical study, single and two-phase models have been used. In single-phase modeling of nanofluid, thermal and flow properties of nanofluid have been considered to be dependent on temperature and volume fraction. Effects of volume fraction and Reynolds number (3000<Re<9000) on convective heat transfer coefficient and pressure drop were investigated for various axial locations of the tube. Numerical results have shown that the inclusion of nanoparticles into the base fluid produced a considerable augmentation of the heat transfer coefficient that increases with an increase of the volume fraction and Reynolds number. Moreover, the increase of volume fraction has no effects on the coefficient of friction, but it decreases with increasing Reynolds number. Comparison of numerical results with experiments shows that the results of single- phase analysis is near to the experimental results.

KEYWORDS: *Heat transfer; Nanofluid; Single Phase/Two-Phase Models; Tube flow; Turbulent Forced convection*

INTRODUCTION

In many industrial applications, the generated heat must enter or leave the system by using a fluid. Therefore, it can be said that the heat transfer phenomenon plays an important role in the industry. Moreover, heat transfer coefficient and thermal conductivity of fluid have an essential role in the efficiency of heat transfer in the equipment such as heat exchangers. With increasing global competition in different industries and the role of energy in the costs of production, these industries tend to the development of novel and improved fluids which have high thermal properties.

Two main areas of thought in the field of nanofluid heat transfer efficiency exist: One of them claims that there is a significant increase in heat transfer coefficient and this does not require consuming a lot of pump power. Another thought claims that increasing the heat transfer coefficient is limited and it is balanced by increasing pumping power [1].

Xuan and Li [2] studied the heat transfer and fluid flow in a pipe by experimental method. Their test results showed an increase in the heat transfer coefficient and Nusselt number with increasing the Reynolds number and nanoparticle volume fraction in turbulence flow**.** Compared to water, in the volume fraction of 2% of copper nanoparticles, Nusselt number increased 39%. Williams et al [3] investigated Al_2O_3 and ZrO_2 nanofluids in water in a horizontal pipe with constant heat flux. They found that in turbulence flow (3000<Re<6300), the Nusselt number can be obtained by Dittus-Boelter equation, when the nanofluid properties are replaced in the equation**.** These results suggest that there is no new mechanism of heat transfer for nanofluids. Rea et al [4] found that in the developing laminar flow using Al_2O_3 and ZrO_2, the measured local Nusselt numbers from experiments are conformed to analytical results. Their results in laminar flow indicated that nanofluids show the behavior of homogeneous mixtures. Fotukian and Esfahany [5] studied experimentally the convective heat transfer and pressure drop of nanofluids in turbulence flow. Their experimental results showed that the flow resistance even at low fractions of nanoparticles is more impressive than the base fluid and experimental results are well predicted by Buongiorno's theoretical equation. Heat Transfer and friction of Al_2O_3 and TiO_2 nanofluids have been investigated experimentally by Pak and Cho [6]. Their experiments showed that the heat transfer coefficient in the volume fraction of 3% is 12% less than the base fluid. However, a dramatic increase in the viscosity can be seen in comparison with the base fluid. Bianco et al [7] numerically studied forced convection of turbulent pipe flow. The single and two-phase models were used for numerical simulation. Their obtained results were in good agreement with the equations. Behzadmehr et al [8] studied forced convection heat transfer in the circular pipe by using single-phase and two-phase models. They reported that the two-phase analysis results are closer to experimental data. Lotfi et al [9] applied the same method to study the forced convective heat transfer in laminar flow.

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Nomenclature

C_p	Specific heat capacity of fluid	T	Temperature
d	Nano particle diameter (m)	U	Inlet velocity
D	Pipe diameter(m)	$\vec{v}_q$	Phase velocity q
$\vec{F}_q$	External volume force	x	Position
$\vec{F}_{vmq}$	Virtual mass force q		**Greek Symbols**
$\vec{F}_{lift,q}$	Lift force for phase q	ε	Turbulence energy dissipation (m^2/s^3)
h_p	Heat transfer coefficient (w/m^2k)	φ	Volume fraction of particles
h_{pq}	Enthalpy of phase	μ	viscosity
H	Enthalpy (j/kg)	μ_q	Shear viscosity of phase q
k	Turbulence energy (m^2/s^2)	ρ	Density ($kg\ m^{-3}$)
K	Thermal conductivity	τ_w	Wall shear stress (N/m^2)
$\dot{m}_{qp}$	Mass transfer from phase p to phase q	λ_q	Bulk viscosity of phase q
Nu	Nusselt number		**Subscripts**
P	Pressure ($= N/m^2$)	B	Bulk
Pr	Prandtl number ($\mu c_p/k$)	f	Index of nanofluids
q"	Heat flux (w/m^2)	m	Index of mixture
Q_{pq}	Intensity of heat exchange between the phase p and q	nf	Index of nanofluids
r,x	Dimensional coordinates	s	Index of surface
$\vec{R}_{pq}$	Interaction between phases	w	Index of wall
Re	Reynolds number	p	Index of particles

Their analysis results show that single-phase and two-phase Eulerian analysis predict lower values for the Nusselt numbers. On the other hand, Yu et al [10] stated that in order to predict the heat transfer coefficient in turbulent flow in low volume fractions, nanofluid can be considered in a homogeneous environment and the theory of single-phase heat transfer can be used for computations.Vahidinia and Rahmdel [11] numerical investigated the turbulent mixed convection heat transfer of the Al_2O_3 water nanofluids in a horizontal circular tube using the two-phase mixture model. Hanafizadeh et al. [12] compared two-phase and single-phase approaches in simulating forced convective heat transfer of Fe_3O_4–water nanofluid in both developing and fully developed regions of a tube under constant heat flux. Macroscopic modeling of hydrodynamic and thermal behavior of nanofluid flows at the entry region of uniformly heated pipe using single-phase and two-phase models has been studied by Goktepe et al[13]. Behroyan et al. [14] compared the predictions of five types of computational fluid dynamics (CFD) models, including two single-phase models (i.e. Newtonian and non-Newtonian) and three two-phase models (Eulerian–Eulerian, mixture and Eulerian–Lagrangian), to investigate turbulent forced convection of Cu water nanofluid in a tubewith a constant heat flux on the tube wall. Forced convection flows of nanofluids consisting of water with Al_2O_3 nanoparticles in a double-tube counter flow heat exchanger have been investigated numerically using single-phase model by Shehnehpour et al [15]. In addition, a lot of researches have used single or two phase model in their studings[16-18]. In the present study, modeling of convective heat transfer in the developing region of turbulent pipe flow containing water- alumina nanofluid with constant wall heat flux was carried out using finite volume method. Single and Eulerian two-phase models were used to approximate behavior of the nanofluid. Effects of volume fraction and Reynolds number on convective heat transfer coefficient and pressure drop were investigated. Although many scientists have investigated multiphase models before but the difference between this studying and others is about the type of heat transfer (forced or natural) and flow (turbulant or laminar). On the other hand, it can be mentioned that up to our best knowledge, none of the papers have studied convection and pressure drop using both phases before.

MATHEMATICAL MODELING

Geometrical Configuration

The geometry of the problem is a pipe of length L = 2.68 m, inner diameter of D = 1.3462 cm and the wall thickness of t = 0.2413 cm which is made of copper. Figure 1 shows the geometrical configuration under consideration. It consists of the steady, forced turbulent convection flow and heat transfer of a nanofluid flowing inside a straight tube of circular cross-section. The fluid enters with uniform velocity and axial temperature profiles at the inlet section. The tube is long enough so that the fully developed conditions prevail at the outlet section. Axially and circumferentially uniform wall heat flux condition has been considered in this treatise. Further, the flow and the temperature fields are assumed to be symmetrical with respect to the vertical plane passing through the tube main axis, thus a two-dimensional flow field can be considered.

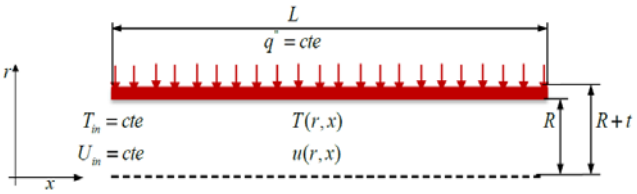


Fig. 1. Schematic diagram of heated circular pipe

Single-Phase Modeling

Assumptions

There exists no formulated theory to date that could reasonably predict the flow behavior of a nanofluid by considering it as a multi-component material. Nanofluids are generally dilute mixtures of solid-liquid that contains very fine particles (smaller than 100nm). Because of such extremely reduced dimension, it has been suggested that these particles may be easily fluidized and consequently, can be considered to behave more like a fluid. Furthermore, by assuming negligible motion slip between the particles and the continuous phase, and the thermal equilibrium conditions also prevail, the nanofluid may be then considered as a conventional single-phase fluid with effective physical properties being function of the properties of both constituents and their respective concentrations. An interesting result from such an assumption resides in the fact that an extension from a conventional fluid to a nanofluid appears feasible, and one may expect that the classical theory as developed for conventional single-phase fluids can be then applied to nanofluids as well. Thus, all the equations of conversation (mass, momentum and energy) as well known for single-phase fluids can be directly extended and employed for nanofluids. In the present study, in conjunction with the aforementioned arguments, we have adopted the 'single-phase' approach to study the thermal behaviors of the nanofluid. For the particular applications under consideration, we have assumed that the nanofluid is incompressible with variable physical properties with temperature. Further, both the compression work and viscous dissipation are assumed negligible in the energy equation.With the above assumptions, the general conversation equations are as follows [19, 20]:

$$div(\rho Vu) = -\frac{\partial P}{\partial x} + div\left[(\mu + \mu_t)gradu\right] + S_u \quad (1)$$

$$div(\rho Vv) = -\frac{\partial P}{\partial r} + div\left[(\mu + \mu_t)gradv\right] + S_v \quad (2)$$

$$div(\rho VT) = div\left[\left(\frac{\mu}{pr} + \frac{\mu_t}{\sigma_k}\right)gradv\right] + S_T \quad (3)$$

$$div(\rho Vk) = div\left[\left(\frac{\mu + \mu_t}{\sigma_k}\right)gradk\right] + G_k - \rho\varepsilon \quad (4)$$

$$div(\rho V\varepsilon) = div\left[\left(\mu + \left(\frac{\mu_t}{\sigma_\varepsilon}\right)\right)grad\varepsilon\right] + c_{1\varepsilon}\left(\frac{\varepsilon}{k}\right)G_k - c_{2\varepsilon}\rho\left(\frac{\varepsilon^2}{k}\right) \quad (5)$$

Where G_k is the perturbation energy production, σ_k is effective Prandtel for turbulence energy and σ_ε is turbulence energy loss. $C_{1\varepsilon}$ and $C_{2\varepsilon}$ are constants and μ_t is perturbation viscosity which is defined as:

$$\mu_t = \frac{\rho C_\mu k^2}{\varepsilon} \quad (6)$$

C_μ is a constant value which equals to 0.09. In Equations 4 and 5, $C_{1\varepsilon}$ =1.44 , $C_{2\varepsilon}$=1.92 , σ_k =1 and σ_ε =1.3.

Physical Properties of the Nanofluids

In single-phase model, thermo-physical and flow properties of nanofluids are predicted by using theoretical models. In this study, the experimental model presented by Massimo Corciono[12] has been applied to predict thermal conductivity and viscosity of nanofluids. These models have been obtained according to the results of the experiments of many researchers by using regression analysis.

$$\frac{k_{nf}}{k_f} = 1 + 4.4\,Re^{0.4}\,Pr^{0.66}\left(\frac{T}{T_{fr}}\right)^{10}\left(\frac{k_P}{k_f}\right)^{0.03}\varphi^{0.66} \quad (7)$$

in which Re is the Reynolds number of nanoparticle and T_{fr} is the freezing point temperature of the base fluid. Reynolds number is defined as follows:

$$Re = \frac{\rho_f u_B d_p}{\mu_f} \quad (8)$$

u_B is the Brownian motion of nanoparticle and is defined as:

$$u_B = \frac{2K_b T}{\pi\mu_f d_P^2} \quad (9)$$

$$\frac{\mu_{nf}}{\mu_f} = \frac{1}{1 - 34.87(d_p/d_f)^{-0.3}\varphi^{1.03}} \quad (10)$$

in which d_f is diameter of the base fluid molecules, which is calculated by the following equation:

$$d_f = 0.1\left(\frac{6M}{N\pi\rho_{f0}}\right) \quad (11)$$

M is the base fluid molecular weight, N is Avogadro's number and ρ_{f0} is the density of the base fluid at the temperature of $T_0 = 293K$. In order to calculate μ_{nf} in each temperature, the value of μ_{nf} should be replaced in the equation. If the base fluid is water, the viscosity of it in each temperature can be obtained as following [21]:

$$\mu_{Water} \times 10^{-4} = exp(\frac{1.12646 - 0.039638T}{1 - 0.00729769T}) \quad (12)$$

In this equation, the viscosity of water is presented in centipoises.

The majority of the researchers agree to use the mixture's law to calculate the density and specific heat of nanofluids**.**The equations are as follows:

$$\rho_{nf} = (1-\varphi)\rho_f + \varphi\rho_p \quad (13)$$

$$(c_p)_{nf} = \frac{(1-\varphi)(\rho c_p)_f + \varphi(\rho c_p)_P}{(1-\varphi)\rho_f + \varphi\rho_P} \quad (14)$$

Two-Phase Method

Although compared to other solid-liquid mixtures with particles larger than micrometers and millimeters, nanofluids behave more like a fluid, they are essentially two-phase fluids and contain some common characteristics of solid- liquids. However, due to the nano-scale size of the particles, applying the conventional theory for two-phase flow of nanofluids may be questionable. Eulerian multi-phase model allows the modeling of multiple distinct phases with the phase interactions. Phases can be liquid, gas or solid or any combination of them. Eulerian view is used for each phase**.**

Boundary Conditions

The governing equations (1-5) constitute a non-linear and coupled system of equation that must be solve subject to appropriate boundary conditions. For the problem under consideration, at the tube inlet, profiles of uniform axial velocity u_0, temperature T_0 prevail. At the tube outlet section, the fully developed conditions are imposed, that is to say that all axial derivatives are zero. On the wall, the no-slip conditions are imposed; also, the thermal boundary condition which has been considered is the uniform wall heat flux. As mentioned earlier, both the flow and thermal fields are assumed to be symmetrical with respect to the vertical plane passing through the tube axis.

For modeling heat transfer and solving the problems in turbulent flow, Simple algorithm has been used. Non-slip boundary condition on the wall for velocity and constant heat flux for energy equation is applied. Heat flux exerted on the wall of the pipe is selected in a way that the maximum temperature of the fluid does not exceed the allowed temperature range for thermal conductivity and viscosity models. The governing equations are as follows:

Mass Conservation

$$\frac{\partial}{\partial t}(\alpha_q\rho_q) + \nabla.(\alpha_q\rho_q\vec{v}_q) = \sum_{p=1}^{n}(\dot{m}_{pq} - \dot{m}_{qp}) + s_q \quad (15)$$

Conservation of Momentum

Momentum balance for phase q leads to the following equation:

$$\frac{\partial}{\partial t}(\alpha_q\rho_q\vec{v}_q) + \nabla.(\alpha_q\rho_q\vec{v}_q\vec{v}_q) = -\alpha_q\nabla P + \nabla.\bar{\bar{\tau}}_q + \alpha_q\rho_q\vec{g} + \sum_{p=1}(\vec{R}_{pq} + \dot{m}_{pq}\vec{v}_{pq} + \dot{m}_{qp}\vec{v}_{qp}) + (\vec{F}_q + \vec{F}_{lift.q} + \vec{F}_{vm.q}) \quad (16)$$

where $\bar{\bar{\tau}}_q$ is the qth phase stress-strain tensor.

$$\bar{\bar{\tau}}_q = \alpha_q\mu_q(\nabla\vec{v}_q + \nabla\vec{v}_q^{\,T}) + \alpha_q(\lambda_q - \frac{2}{3}\mu_q)\nabla.\vec{v}_q\bar{\bar{I}} \quad (17)$$

Lift Force

This force is generally the result of velocity gradient in the flow field of the first phase. Lift forces are more important for the bigger particles. Lift force exerted on the second phase of p in the first phase of q is calculated by:

$$\vec{F}_{lift} = -0.5\rho_q\alpha_q(\vec{v}_q - \vec{v}_p) \times (\nabla \times \vec{v}_q) \quad (18)$$

Lift force will be added to the right side of the momentum equations for both phases.

Virtual Mass Force

For multi-phase flows, the effect of the virtual mass force appears when the secondary phase, P is accelerated compared with initial phase momentum, q. The mass inertia of the initial phase because of the particle acceleration, is exerted as the virtual mass force on a particle:

$$\vec{F}_{vm} = 0.5\alpha_q\rho_q(\frac{d_q\vec{v}_q}{dt} - \frac{d_p\vec{v}_p}{dt}) \quad (19)$$

Term $\frac{dq}{dt}$ denotes the time derivative of the phase and is as follows:

$$\frac{d_q(\phi)}{dt} = \frac{\partial(\phi)}{\partial t} + (\vec{v}_q.\nabla)\phi \quad (20)$$

The virtual mass force will be added to the right side of momentum equations for both phases. Virtual mass effect is important when the secondary phase density is much smaller than the primary phase density.

Conservation of Energy

To explain the conservation of energy in multi-phase Eu-

-lerian applications, a separate enthalpy equation can be written for each phase:

$$\frac{\partial}{\partial t}(\alpha_q \rho_q h_q) + \nabla.(\alpha_q \rho_q \vec{v}_q h_q) = -\alpha_q \frac{\partial p_q}{\partial t} + \overline{\overline{\tau}}_q : \nabla \vec{u}_q - \nabla.\vec{q}_q + s_q + \sum_{p=1}^{n}(Q_{pq} + \dot{m}_{pq} h_{pq} - \dot{m}_{qp} h_{qp}) \quad (21)$$

Turbulence Modeling In Two-Phase Model

Compared with the single-phase, the terms of momentum equation in multi-phase flows is more and this is due to the simulation of complex turbulence model of multi-phase flows. K-ε turbulence model has been used in this study. This model is actually a generalization of the single-phase K-ε model.

In this case, the properties of the mixture and mixture velocity are sufficient to obtain the flow characteristics k and ε equations are as follows:

$$\frac{\partial}{\partial t}(\rho_m k) + \nabla.(\rho_m \vec{v}_m k) = \nabla.(\frac{\mu_{t,m}}{\sigma_k}\nabla_k) + G_{k,m} - \rho_m \varepsilon \quad (22)$$

$$\frac{\partial}{\partial t}(\rho_m \varepsilon) + \nabla.(\rho_m \vec{v}_m \varepsilon) = \nabla.(\frac{\mu_{t,m}}{\sigma_k}\nabla_\varepsilon) + \frac{\varepsilon}{k}(C_{1\varepsilon} G_{k,m} - C_{2\varepsilon}\rho_m \varepsilon) \quad (23)$$

where the mixture density ρ_m and velocity $\vec{v}_m$ are calculated by:

$$\rho_m = \sum_{i=1}^{n} \alpha_i \rho_i \quad (24)$$

$$\vec{v}_m = \frac{\sum_{i=1}^{n} \alpha_i \rho_i \vec{v}_i}{\sum_{i=1}^{n} \alpha_i \rho_i} \quad (25)$$

Turbulence viscosity $\mu_{t,m}$ is obtained using the following equation:

$$\mu_{t,m} = \frac{\rho C_\mu k^2}{\varepsilon} \quad (26)$$

And the production of turbulence kinetic energy is as follows**:**

$$G_{k,m} = \mu_{t,m}(\nabla \vec{v}_m + (\nabla \vec{v}_m)^T) : \nabla \vec{v}_m \quad (27)$$

Constants in these equations are:

$$C_{1\varepsilon} = 1.44,\ C_{2\varepsilon} = 1.92,\ C_\mu = 0.09$$
$$\sigma_k = 1.0,\ \sigma_\varepsilon = 1.3 \quad (28)$$

Grid Study

The governing non-linear equations have been solved by using the finite volume method and Sample algorithm. Exponential rule used for the transportation and distribution terms. In order to check the grid independence of the obtained solutions, governing equations were solved using multiple grids.

The effect of grid on various parameters was studied carefully. For example, in Figure 2, the axial velocity profile at x=1(m) is shown for both single-phase and two-phase methods. It is observed that when the numbers of nodes are more than 50 × 280, increasing the number of nodes would have no effects on the solution. Therefore a 50 × 280 grid has been chosen to solve the problem.

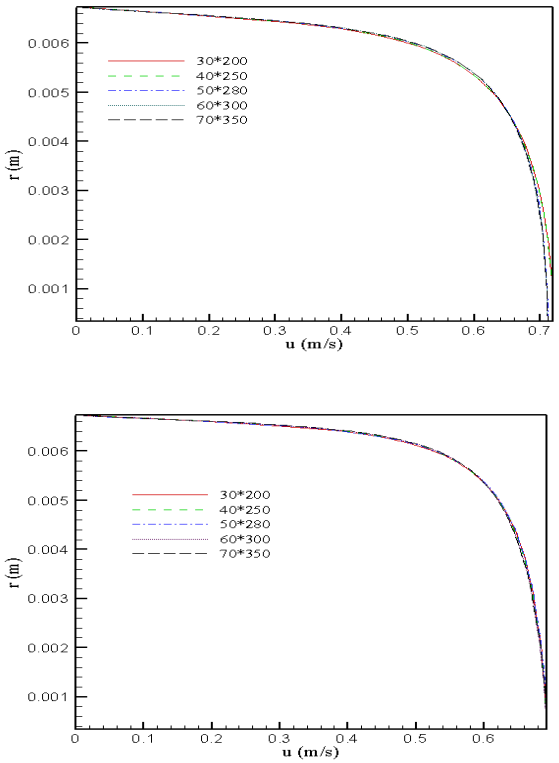


Fig. 2. The effect of the number of grids on the axial velocity profile at x= 1 m. A: single phase method- B: two phase method

RESULTS

In order to verify the accuracy of the simulation, the results have been compared with the experimental results of Yang[1].

In Figure 3, Nusselt number obtained from the numerical solution in the volume fraction of 2.55% is compared with the experimental results of Yang. It can be seen that the results of the single-phase analysis is closer to the experimental results. The two-phase model's results predict higher amounts for Nusselt number in a certain Reynolds number.

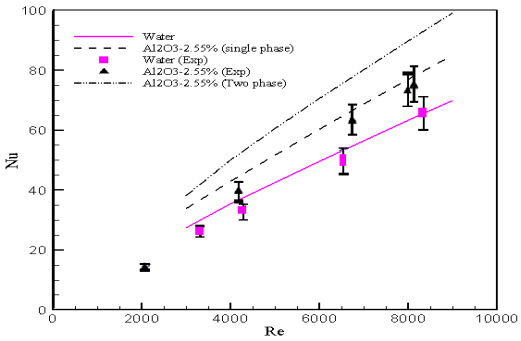


Fig. 3. Comparison of the Nusselt number obtained from numerical analysis with experimental results of Yang

In Figure 4, the pressure drop in the pipe is compared with the experimental results of Yang. According to this figure, it is clear that the single-phase analysis provides better results of flow field. Both single-phase and two-phase methods, predict lower pressure drop compared to the experimental results but the difference in single-phase is much less. The results are also in good agreement with the results of Yu et al [10].

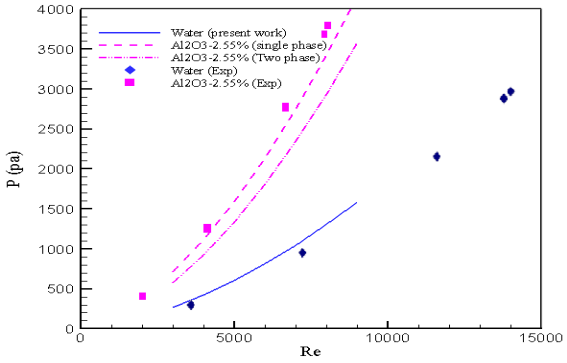


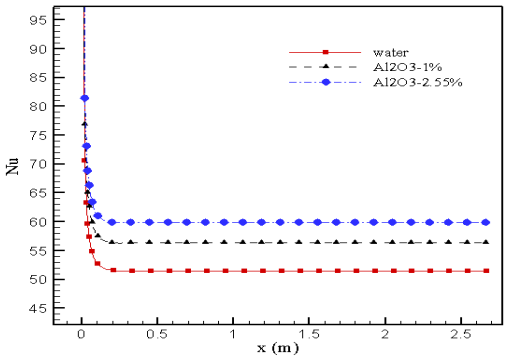
Fig. 4. Comparison of the pressure drop obtained from the numerical analysis of the experimental results of Yang

In Figure 5, Nusselt number and heat transfer coefficient is shown along the pipe at Re=6000. According to the figures, the exerted heat flux to the pipe is q"=26332.09 (W/m^2). It is observed that the flow reaches the state of development immediately after entering the pipe. Moreover, increasing the volume fraction of alumina nanoparticles increases the Nusselt number and the heat transfer coefficient. Nussel number is defined as follows:

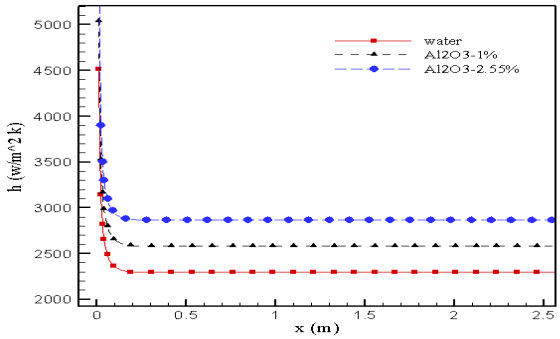
$$h = \frac{-K\dfrac{\partial T}{\partial r}}{T_s(x) - T_b(x)} \tag{29}$$

$$Nu = \frac{hd}{K} \tag{30}$$

$$Re = \frac{\rho UD}{\mu} \tag{31}$$



(a)



(b)

Fig. 5. The effect of increasing of volume fraction of nanoparticles on the local Nusselt and local heat transfer coefficient at Re=6000

In Figure 6, the average Nusselt number obtained by the single-phase method is plotted based on Reynolds number. The diagram indicates that the Nusselt number increases with increasing Reynolds number and nanoparticle volume fraction. In this figure, the Nusselt number is drawn using Dittus- Boelter and Gnlienski theoretical equations for base fluid of water. Nusselt number values resulted from the numerical solution for water, are very near to the amounts predicted by the two models. Dittus- Boelter equation is as follows [22]:

$$Nu = 0.023\, Re_D^{0.8}\, Pr^{0.4} \tag{32}$$

Gnlienski relationship is as follows [22]:

$$Nu_D = \frac{(f/8)(Re_D - 1000)\,Pr}{1 + 1.27(f/8)^{1/2}(Pr^{2/3} - 1)} \tag{33}$$

The friction coefficient can be calculated from the following equation [22]:

$$f = (0.79 Ln\, Re_D - 1.64)^{-2} \tag{34}$$

By solving the problem, the coefficient of friction in the pipe is obtained which is shown in Figures 7 and 8. The

results display that the friction coefficient decreases with increasing Reynolds number but volume fraction increase, causes no significant change in the average friction coefficient of pipe.

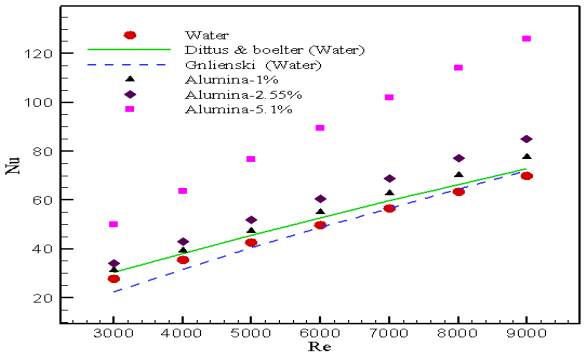


Fig. 6. Average Nusselt number varying with Reynolds number using single-phase method

The coefficient of friction is defined as follows:

$$\tau_w = \mu(\frac{\partial u}{\partial r})_{r=r_w} \quad (35)$$

$$c_f = \frac{\tau_w}{\rho U^2} \quad (36)$$

Where μ and ρ are the fluid viscosity and density, respectively.

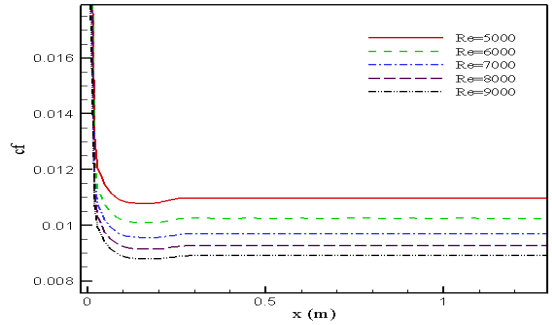


Fig. 7. Changes in friction coefficient versus the length of pipe for different Reynolds numbers

Pressure drop versus Reynolds number in the pipe at various volume fractions is indicated in Figure 9. The diagram shows the pressure drops sharply increases with increasing volume fraction of nanoparticles.

The reason is due to this fact that the viscosity and density of nanofluid have been increases compared with the base fluid.

DISCUSSION

In this paper, the forced convective heat transfer in water-alumina nanofluid has been studied numerically using single-phase and two-phase methods. In the single-phase method, viscosity and thermal conductivity of nanofluid is varying with temperature and volume fraction of nanoparticles. Comparison of numerical results obtained from the numerical solutions with experimental results show that the results of single-phase analysis are closer to the experimental results.

It was also observed that with increasing volume fraction and Reynolds number, Nusselt number and heat transfer coefficient increases. Furthermore, the results show that by increasing Reynolds number, pressure drop increases, which is greater at higher volume fractions. Friction coefficient decreases with increasing Reynolds number and volume fraction changes have negligible effects on the friction.

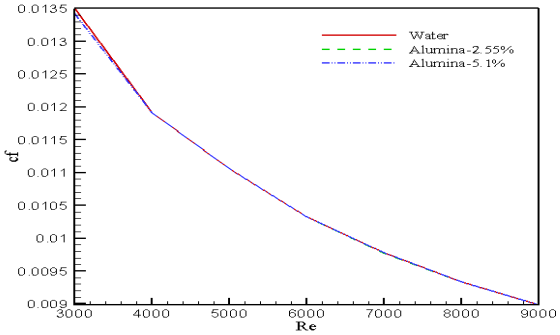


Fig. 8. Changes in average friction coefficient versus Reynolds number

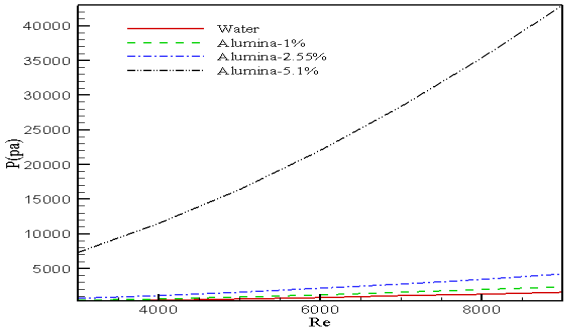


Fig. 9. Pressure drop in the pipe versus Reynolds number

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